BOOSTED AIR SOURCE HEAT PUMP

BACKGROUND

This invention relates to air source heat pumps, and especially to air source heat pumps suitable for use in normally colder climates. This invention is a modification and improvement to the inventions in prior U. S. Patents 6,276,148 and 5,927,088 in that this invention presents a somewhat simpler and less expensive air source heat pump system. The entire contents of my prior U. S. Patents 6,276,148 and 5,927,088 are incorporated herein by reference.

Heat pump capacity is understood in the art to be the amount of heat per unit time delivered by a heat pump system. An effective heat pump system must deliver adequate capacity with low or falling outdoor temperature, while avoiding the delivery of too much capacity when the outdoor temperature warms up. Heat pumps that are designed to deliver sufficient capacity at very cold outdoor temperatures may have the inherent problem of delivering too much capacity when the outdoor temperature warms up, because more energy is available to be obtained from the warmer outdoor air. This may cause system inefficiencies as the system heat exchangers become overloaded and compressor power consumption rises to inefficient levels. In extreme cases, safety controls may cause the compressor to stop running.

My prior U.S. Patents 6,276,148 and 5,927,088 disclose systems for adjusting capacity in response to decreases and/or increases in outdoor ambient temperature. This involves primary and booster compressors connected in series and a microprocessor which responds to a sensed parameter commensurate with outdoor ambient air temperature to allow sequences of compressor operation for capacity levels consistent with heating and/or cooling requirements and consistent with efficient and safe operation of the system. While the systems of my prior patents are effective for their intended purpose, and particularly for use in very cold climates, a need exists for a simpler and less expensive system.

SUMMARY OF THE INVENTION

As with prior U.S. Patents 6,276,148 and 5,927,088, the present invention employs a primary and a booster compressor in series. In the simplest embodiment of this invention, both compressors may be single speed compressors. For the primary compressor, however, at least a two speed or unloadable compressor is preferred

Heating and cooling system control is achieved with a multi-step indoor thermostat employed in conjunction with a refrigerant system low side pressure sensor that is commensurate with outdoor ambient temperature. The multi-step thermostat has settings that call for various steps of heating or cooling operation; and the low side pressure sensor operates to prevent unnecessary, unsafe and/or inefficient operation of the system. The complication and expense of a microprocessor can be eliminated.

The present invention also includes a circulating defrost energy transfer fluid system (as defined herein) for a defrost cycle for the heat pump mode of operation. Instead of having to use resistance heating while pulling heat out of the heated indoor air for the defrost cycle, the present invention utilizes a defrost energy transfer fluid for heat exchange to add thermal energy to vaporize the circulating refrigerant to effect the defrost operation. This eliminates the necessity of using electric resistance heating, thereby avoiding cold drafts during the defrost cycle.

BRIEF DESCRIPTION OF THE DRAWINGS

Referring now to the drawings, in which similar elements are numbered alike in the several figures:

Figure 1 is a schematic diagram of the heat pump system of the present invention in the lowest heating capacity modes of operation where only the primary compressor is operating.

Figure 2 is a schematic diagram of the heat pump system of the present invention in the next higher heating capacity mode of operation in which both the primary and booster compressors are operating.

Figure 3 is a schematic diagram of the heat pump system of the present invention in the next higher heating capacity mode of operation in which the primary and booster compressors and economizer are all operating.

Figure 3a is a schematic diagram of the heat pump system similar to Figure 3, but incorporating a plurality of pressure sensors for various control purposes.

Figure 4 is a schematic diagram of the heat pump system of the present invention in a cooling capacity mode of operation in which just the primary compressor is operating (either one speed, two speed or unloadable).

Figure 5 is a schematic diagram of the heat pump system of the present invention in a defrost mode of operation and with a water defrost system.

Figure 6 is a table showing various possible heating and cooling modes of operation for the system of the present invention.

DESCRIPTION OF THE PREFERRED EMBODIMENT

Referring to Figures 1, 2 and 3, a closed loop heat pump system is shown. The system has a booster stage compressor 22, a primary compressor 24, an indoor coil or condenser 26 which delivers heat to a space to be heated, an outdoor coil or evaporator 28, and conduit means 29 connecting these elements in a closed loop circuit. The system also has a refrigerant system pressure sensor 30, the output of which is commensurate with outdoor ambient air temperature. Pressure sensor 30 is located in a position so that it is always exposed to the refrigerant low pressure side of the system. The heat pump system also has a four way valve 32 to reverse the direction of refrigerant flow when the system switches from the heating

mode to the cooling mode, and vice-versa; an indoor thermostat 34 to select and sense the temperature of the air in the space to be heated (or cooled); and a control panel/mini-microprocessor 36 for receiving signals from the thermostat and pressure sensor 30 and for interpreting those signals and directing electric power to system components as required. The foregoing are the basic components of the closed loop heat pump system.

Booster compressor 22 and primary compressor 24 are connected in series when both are operating. Both compressors may be single speed and single capacity units. However, it is preferred that primary compressor 24 be at least a two speed or an unloadable compressor. For the purposes of this invention, the term "multi-capacity compressor" will mean any compressor that has more than one speed or that is unloadable.

Indoor thermostat 34 is any of the commercially available thermostats that are capable of multiple steps for calling for heating and/or cooling steps. In its simplest configuration, the thermostat will be designated as T1 and will have two heating steps, H1 and H2, and one cooling step, C1, (such as a T8511G 1021 available from Honeywell; it may also be a thermostat designated as T2 having two heating steps, H1 and H2, and two cooling steps, C1 and C2, (such as T8511M 1002 available from Honeywell; or it may be a thermostat designated T3 having three heating steps, Hi, H2 and H3, and two cooling steps, C1 and C2, (such as T8611M 2025 available from Honeywell).

In the following discussion, heating operation will be discussed first, and then cooling operation will be discussed.

Figure 1 shows the system in its simplest mode of heating operation. Thermostat 34 is calling for the first step of heating operation. If both primary compressor 24 and booster compressor 22 are single speed and non-unloadable compressors, the signal will be sent from thermostat 34 to control panel 36 to deliver electric power to operate primary compressor 24 full on. Check valve 38 will be open, and booster compressor 22 will be off. This is indicated as Mode 2 in line T1 of Figure 6, where T1 is a thermostat type having two heating steps and one cooling step, and the system is operating at H1. (This system does not operate in Mode 1,

because the primary compressor is not capable of partial capacity operation.) The path of refrigerant flow is as shown by the arrows from primary compressor 24, through four way valve 32, to indoor coil 26, bypassing expansion valve 27, to expansion valve 40 and outdoor coil 28, and then through four way valve 32 back to the inlet to primary compressor 24. Air handlers and/or fans for indoor coil 26 and outdoor coil 28 are not shown.

Referring to Figure 2, if the system capacity is insufficient to satisfy thermostat 34, the thermostat will call for the second heating step H2. This will result in control panel 36 sending electric power to operate booster compressor 22 full on. This is indicated at line T1, Mode 3 in Figure 6, where the system is operating at H2. Check valve 38 is closed, and refrigerant flow will then be from four way valve 32 to booster compressor 22, and then from the booster compressor 22 to primary compressor 24, and then through the system as described above.

Thermostat T2 in Figure 6 has two heating and two cooling steps. As indicated on the T2 line of Figure 6, Modes 2 and 3, the heating operation with this thermostat is the same as for thermostat T1. However, for the two cooling steps of the T2 thermostat, the primary compressor 24 is a multi-capacity compressor to accomplish the two cooling steps.

Several modes of heating operation are possible with a T3 thermostat. For all T3 heating modes, primary compressor 24 is a multi-capacity compressor. As indicated in the first T3 line in Figure 6, Modes 2 and 3 can be the same as for a T1 or T2 thermostat. That is, the first heating step, H1, results in primary compressor 24 being full on (booster off) and the second heating step, H2, results in adding booster compressor 22 full on, so that both compressors are full on. If system capacity is insufficient to satisfy the thermostat T3, the third heating step of the thermostat, H3, will result in adding electric resistance or other back-up heating 31 to the system (Mode 5).

Referring to the second T3 line in Figure 6, in Mode 1 the first heating step, H1, results in operation of primary compressor 24 at partial capacity. In Mode 2, heating step H2 results in full capacity operation of primary compressor 24. In Mode 3, heating step H3 results in

adding booster compressor 22 to the system, so that the system has the heating capacity of the booster and the full primary compressors.

Referring both to Figure 3 and the third T3 line in Figure 6, an economizer 42 is added to the heating system. At Mode 2, heating step H1 causes full capacity operation of primary compressor 24; and at Mode 3, heating step H2 adds the operation of the booster compressor. At Mode 4, heating step H3 results in activation of the economizer. This is effected by delivery of an electrical signal from control panel 36 to solenoid 44 to bleed some of the liquid refrigerant to the boiling side of the economizer, from which that bled refrigerant is then delivered to an interstage location between the booster and primary compressors after being vaporized in the economizer. Operation of the economizer is more fully discussed in my prior patents 6,276,148 and 5,927,088.

The fourth T3 line in Figure 6 represents a system in which electrical resistance or other back-up heat is added to the system of Figure 3. As indicated in the fourth T3 line of Figure 6, this system is configured so that the H2 heating step calls for both booster compressor 22 and economizer 42 to be activated at Mode 2 in response to heating step H2. At Mode 5, if the heating capacity of the system is not sufficient to satisfy the thermostat, the heating step H3 adds electric resistance (or other backup) heat to the system.

While backup resistance heat 31 is called for or operated at Mode 5 for the first and fourth T3 lines of figure 6, it will also be understood that backup resistance heat can be manually selected with any of the thermostats.

Rate of change thermostats are also now available where the thermostat steps are activated when the rate of change of the temperature of the indoor air being heated (or cooled) does not meet a predetermined standard. Such rate of change thermostats can also be used with the present invention, provided they have at least two heating steps and one cooling step.

Cooling operation and/or defrost operation of the system is shown in Figure 4. Four way valve 32 is repositioned so that the refrigerant discharged from primary compressor 24 is

delivered, as shown by the flow arrows, to outdoor coil 28, which is now functioning as a condenser, and then around expansion valve 42 via the now open associated check valve. The refrigerant then flows through expansion valve 27 (since the associated check valve is closed) to coil 26, which is now functioning as an evaporator, and then back to the four way valve 32 and back to the inlet to primary compressor 24. In the system of the present invention, booster compressor 22 does not operate on the cooling cycle. There is either one cooling step, C1, where primary compressor 24 is a one speed or non-unloadable compressor operated at full capacity (Mode 2), or two cooling steps, where primary compressor 24 is a multi-capacity compressor, with the first cooling step, C1, being at partial capacity operation of the primary compressor (Mode 1), and the second cooling step, C2 being at full capacity operation of the primary compressor (Mode 2).

Line T1 of Figure 6 reflects a system with one cooling step. In Mode 2, the thermostat calls for the single cooling step C1, which results in full capacity operation of primary compressor 24. As also shown in Figure 6, for the T2 thermostat and all T3 thermostats, cooling step C1 called by the thermostat results in Mode 1 cooling operation where primary compressor 24 is operated at partial capacity. If the cooling capacity of the system in Mode 1 does not satisfy the thermostat, cooling step C2 will be called by the thermostat, resulting in Mode 2 operation where primary compressor 24 is operated at full capacity.

It will be understood that the heating and cooling sequences of operation shown in Figure 6 are by way of illustration. Other heating and cooling sequences may be effected in conjunction with the particular components (compressors, economizer, backup heat, etc.) used in a system.

An important feature of the present invention is the incorporation of pressure sensor/transducer 30 in the refrigerant flow line upstream of the inlets to both primary compressor 24 and booster compressor 22. Pressure sensor 30 senses the pressure of the refrigerant upstream of the inlet to the compressors, i.e., approximately the system low side or suction pressure, which pressure is commensurate with outdoor ambient air temperature during all heating cycle modes of operation. That is, system low side pressure will rise with

rising outdoor ambient temperature, and will fall with falling outdoor ambient temperature. Pressure sensor 30 delivers an electrical signal to a mini-microprocessor (preferably located in control panel 36), which serves to prevent outputs from control panel 36, which would otherwise occur in response to signals from indoor thermostat 34, to do some or all of the following in the heating mode of operation of the heat pump:

1. Prevent operation of booster compressor 22 whenever the system low side pressure is higher than a first predetermined level.
2. Prevent operation of economizer 42 whenever the system low side pressure is higher than a second predetermined level.
3. Prevent operation of electric resistance heat whenever the system low side pressure is higher than a third predetermined level.
4. Prevent partial operation of primary compressor 24 whenever the system low side pressure is below a predetermined level
5. Act as a system safety shutdown whenever the system low side pressure is below a predetermined level during all heating and cooling modes, which condition indicates, e.g., a loss of refrigerant charge.

Instead of one pressure sensor and a microprocessor to accomplish some or all of the control features 1-5 itemized above, these features can be accomplished by employing a plurality of individual pressure sensors, all of which are positioned upstream of the inlet to either the primary compressor or the booster compressor to sense approximately system low side or suction pressure. The use of a microprocessor is then not required. Figure 3a shows the system with six individual pressure sensors, PS1 – PS 6, all of which are positioned to sense approximately the system low side or suction pressure.

PS 1 is connected to operate a normally open switch in the booster power line. PS 1 closes its associated switch when the sensed pressure falls to a preset point, thereby permitting operation of booster compressor 22 if called for by the indoor thermostat.

PS 2 is connected to operate a normally open switch in the power line to the economizer solenoid. PS 2 closes its associated switch when the sensed pressure falls to a preset point, thereby permitting operation of economizer 42 if called for by the indoor thermostat.

PS 3 is connected to operate a normally open switch in the power line to backup resistance heat 31. PS 3 closes its associated switch when the sensed pressure falls to a preset point, thereby permitting operation of backup heat 31 if called for by the indoor thermostat.

PS 4 is connected to operate a normally closed switch in the power line to primary compressor 24. PS 4 opens its associated switch when the sensed pressure falls to a preset point, thereby preventing operation of primary compressor 24 and acting as the system low pressure safety control when the system is operating in the cooling or defrost modes.

PS 5 is connected to operate two normally closed switches in the power lines to primary compressor 24 and booster compressor 22. PS 5 opens its two associated switches when the sensed pressure falls to a preset point, thereby acting as a system low pressure safety control by interrupting or preventing operation of both the primary compressor and the booster compressor when the sensed pressure indicates a safety condition, such as loss of refrigerant charge, when the system is operating in the heating mode.

PS 6 is connected to cause full capacity operation of the primary compressor 24 when the sensed pressure falls to a preset point, even if a particular thermostat is calling for partial capacity operation of the primary compressor. This prevents system operation at too low a refrigerant mass flow due to a thermostat calling for partial capacity operation of the primary compressor when outside air temperature is too cold for safe partial capacity operation of the primary compressor.

The switches associated with each of the pressure sensors PS 1 –PS 6 are preferably located in control panel 36.

As is well known in the art, heat pump operation in the heating mode below a predetermined outdoor ambient temperature requires the use of a defrost cycle to remove frost or ice from evaporator coil 28. The defrost cycle may be initiated by a timing cycle or a demand measurement. In the timing cycle, defrost operation is initiated at set time intervals, whether needed or not. In the demand cycle, defrost operation is only on demand when called for by the presence of excessive frost/ice on coil 28. There are various direct and indirect system parameter measurements well know in the art that can be utilized to initiate a defrost cycle. Defrost is accomplished by moving four way valve 32 from the Figure 1 position to the Figure 4 position and adding thermal energy to vaporize the circulating refrigerant which is subsequently compressed for delivery to outdoor coil 28. In the prior art, heat for heating the refrigerant is typically obtained from direct cooling of the indoor air that was just previously being heated. Backup electric resistance heat is typically used to prevent cold drafts in the indoor air from which heat is being extracted for defrost.

The use of backup resistance heat to prevent cold drafts during a defrost cycle is expensive. It can also impose an electrical load requirement (e.g. 200 amp service) that many older residential systems (typically 100 amp capacity) may not be able to meet; and retrofitting for 200 amp service may be very expensive or not practicable. This is a problem of particular concern in larger cities having older multi-unit apartment or condo units. The present invention addresses that problem by using a defrost energy transfer fluid as the heat source for the defrost cycle. As used herein, the term "defrost energy transfer fluid" is defined to mean either water, such as the ordinary water supply to a building, or an available waste water source, or a non-freezing or low temperature freezing heat transfer fluid such as any suitable antifreeze.

The defrost energy transfer fluid feature for defrost is shown in Figure 5. Assuming that the heat pump system has been operating in a heating mode, such as in any of Figures 1, 2 or 3, the four way valve will be moved to the cooling mode of operation, where the refrigerant will flow as indicated by the arrows in Figure 5. A solenoid operated valve 50 is closed to prevent refrigerant flow to indoor coil 26; and a solenoid valve orifice 52 is opened so that flashed liquid refrigerant now flows through a bypass line 29a to a heat exchanger 54 where it is in

heat exchange relationship with a defrost energy transfer fluid flowing in line 56. The liquid refrigerant from outdoor coil 28 will thus be evaporated by energy absorbed from the defrost energy transfer fluid in heat exchanger 54. The resulting refrigerant vapor will then flow through four way valve 32 to primary compressor 24 where it is compressed and heated during compression, and then again through the four way valve to outdoor coil 28 to defrost the outdoor coil. As indicated above, if water is the defrost energy transfer fluid, the water may be from any convenient source, such as the normal potable or waste water sources for the building or structure being heated by the system.

While preferred embodiments have been shown and described, various modifications and substitutions may be made without departing from the spirit and scope of this invention. Accordingly, it is to be understood that the foregoing invention has been described by way of illustration and not limitation.